REMARKS

Thesis to show that "orienting the elongated pads at 90 degrees to the radial", as proposed in my (Cherian) invention, produces an unobvious result and advantage, in the view of the practitioner.

The following thesis is intended to show that my (Cherian) way of orienting the elongated attachment pads (pads) does produce an unobvious result and a technical/ utility advantage, which was not anticipated by Washino in his US Patent # 5,484,963, and would not have been obvious to one having ordinary skill in the art at the time the invention was made.

Definitions:

Washino in his US Patent # 5,484,963 shows and claims his elongated pads with their **Long Axes** to be **radial**, extending from a predetermined point. I will refer to his elongated pads orientation as the **Washino "radial"** orientation, wherein the **Long Axis** of each of his elongated pads is extending **radially** from that predetermined point.

In my invention, I am orienting my (Cherian) elongated pads at 90 degrees to Washino's pads orientation. I will refer to my pads orientation as the **Cherian "transverse"** orientation or simply the **transverse orientation** (1). We can describe this (Cherian) orientation at least in one of the following additional ways: 2) In my (Cherian) transverse orientation, the **Long Axis of my** (Cherian) **elongated pads** is **at 90 degrees to Washino's elongated pads orientation**. 3) We can also describe my **(Cherian) transverse** pad orientation as where the long axis is oriented **at 90 degrees to a ray**, wherein said ray extends from the predetermined point to the center of the pad, when said pad is viewed in plan view. 4) We can also describe it as where the **Short Axis** of my (Cherian) elongated pad is extending **radially** from that predetermined point, in contrast to Washino's Long Axis being so.

In any of these definitions, we can always add the word "substantially", to indicate that the direction can vary a few degrees from the theoretical and would still be considered as described.

In the following thesis, whenever I will use any one of these various descriptions, I will mean the same thing. They will all mean that my pads are (substantially) at 90 degrees to Washino's pads. But the best "short hand" description or definition is to say that my elongated pads are oriented in a **"transverse"** orientation **to the rays** extending from that predetermined point.

The purpose of my invention is to counteract the ill effect of thermal fluctuations that may exist and may affect electronics systems or assemblies, and especially the ill effect on the joints between the components of such assemblies. And I am doing that by orienting my elongated pads in the "transverse" orientation, as proposed in my invention. This is a novel way, because it does result in an important technical and utility advantage, as will be explained below.

It is know that when an electronic assembly comprises a number of components, and any two such components are joined together and then they expand or contract at different rates, either due to differences in their individual/respective temperatures or due to a difference in their thermal coefficient of expansion (TCEs), then the joints between these two components get stressed due to the **relative deformation between these two components**. This is because one of the two components becomes longer or shorter than the other one, in a radial direction, the direction of their thermal expansion and contraction, and it then either pushes or pulls on the joints between these two components and the joints get stressed.

If the components are not fixated at any specific point, then the components expand and contract relative to a point, generally/substantially near the geometric center of the two components, assuming that the components are substantially homogeneous in their materials

and construction. This geometric center would also be generally near the geometric center of the group of joints between these two components. I refer to such a point as the thermal center of these two components. If on the other hand, the two components are fixated at a specific point, say for example, one of their respective corners is pushed against a corner at the intersection of two perpendicular registration walls, which would restrict the movement of the corner of the two components, then this corner would be considered the fixation point and it would become the "predetermined point", and all the deformation will relate to that fixation point.

The effect of such a condition, i.e. thermal deformation, is that each joint will be pushed or pulled in the direction of a ray, extending from the thermal center or the fixation point to the joint itself. We can refer to that direction as the radial deformation direction or simply **the "radial" direction.**

Now, we can have at least two different geometry situations, depending on the shape, size and height of the joints; and each situation will affect in a different way the stresses on/in the joints and on the pads. I will refer to these two situation as, A) the "Short or Stubby Joints" and B) the "Tall Joints or Columns." Then I will explain how these two situations affect the stresses on/in the joints and on the pads and the resulting failure mechanism.

A) The "Short or Stubby Joints"

If the joints are short and stubby, and the space or distance or gap between the two joined components is small, then the dominant stresses in the joints will be the "shear" stresses. The direction of the stresses will be substantially in the "radial" direction, which, as explained above, will be in the direction of a ray extending from the predetermined point (the thermal center or the fixation point) to the joint itself. Another effect is that the joint may undergo a "rolling" aspect, which creates what is usually referred to as "torque" or "torsion", as seen from the side view of the joint.

If the thermal deformation is large enough, then the shear stresses on/in the joint and on the pads can create some failures in the joint at the locations of the highest stresses. This shear stress, especially if combined with the torque stress, can be detrimental, and if it is large enough and/or repeated often enough, can reach a level which could ultimately lead to the failure of the joint and could crack/delaminate/ separate the joint from the pad and/or separate/delaminate the pad from the substrate, and obviously could ultimately lead to the failure of the electronic assembly containing this joint.

Can we mitigate this problem? Yes, to some degree, as shown below, by at least two alternatives.

Alternative #1

We could reduce the level of the shear stresses in/on the joints and on the pads by making the joint larger in one or two directions. One by increasing the height of the joint as described later, or two, by making the area of the pad larger, as seen in a plan view. This means, that we would make the "area" of the joint, and the attachment pad, larger. Either by making the attachment pad to have a larger diameter or larger square or larger rectangle. This is because the shear stress is governed by the following mathematical formula:

Shear Stress = Ss = F / A **(equation 1)**, where

Ss = Shear Stress;

F = Force, applied on the body to create the shear (Shear Force);

A = the Area of the pad at the base of the joint.

We can also consider the cross-sectional area of the joint at any specific height or distance from the pad and calculate the shear stress at that cross section.

The shear force gets generated from the fact that the components are pushing against each other due to the fact that one of them is getting longer or shorter than the other due to their temperatures and/or due to their TCE differences.

By analyzing equation 1, we can see that we cannot control the Force F, because the force is controlled by the nature and material of the components and by their geometry and temperatures. So, the only factor in the equation that we can readily control and change is the Area A.

This means that we could increase A, to reduce Ss.

But the electronics industry would not like that. The trend is to "miniaturize" the components and the systems and to make things smaller and smaller all the time.

So this option is not really a good option.

Alternative #2

B) The "Tall Joints or Columns."

This is the other alternative. This situation would occur when the distance between the two joined components is large, larger than in the A) situation described above.

We can obtain this "Tall Joint or Columns" situation, if we deliberately plan to achieve it. One way is to use "pins" like with the "Pin Grid Array Packages"; or when we use the columns described in my three US Patent #4,664,309 "Chip Mounting Device, or US Patent #4,705,205 "Chip Carrier Mounting Device" [CCMD], or US Patent #4,712,721 "Solder Delivery System"; or the "No-Wick" Columns described in my recent US Patent #6,884,707 "INTERCONNECTIONS". I have described the effect of such tall columns in several of the references cited in the specification of my present application and in the specification itself.

But I will go through some engineering/mathematical formulas here below to highlight the effect of the orientation of the columns with elongated cross-sections. **By making the columns cross section rectangular or generally elongated, and by using my**

orientation, we obtain an even greater advantage to simply using the columns, as will be seen below.

The purpose and advantage of using tall columns joints is because this allows us to control and reduce the stress in the joints and on the pads, and thus extend their operating life and improve their reliability of electronic systems. Here is why this can be possible.

When the joint is tall and looks like a columns, and the deformation between the two joined components tries to push or pull the joints, then the column "bends" and acts as a "flex joint". In such a case, the shear stresses get lowered and the dominant/prevailing stresses become the "bending stresses", instead of the shear stresses that were dominant in the short joints described in A) above.

Now, the benefit of converting the stresses to "bending" stresses is that we have more leeway to control and reduce these stresses, as I will explain.

I will first analyze the columns in general, and then I will proceed to compare columns with elongated cross sections in two opposing orientations, i.e. like in Washino's radial orientation and like in my (Cherian) transverse orientation.

I would like to use the reference book listed here below, to obtain the engineering/ mathematical formulas governing these situations.

A good way to analyze the stresses in a column in this arrangement, like those encountered in electronic assemblies, like in our case, is to treat the column as a "beam", that is fixed at both ends (at both pads) and is being pushed or pulled at end one with respect to the other end, i.e. to visualize it as a "beam under loads", as covered by **case 17** in the Reference book.

Another way to analyze the stresses in the column in this arrangement is to consider only one half of the column (half its height), and to consider each half of the column halves as a cantilever beam, fixed at the pad, and being pushed or pulled at its free end (mid height point of the original column) with respect to the fixed pad end. This is covered by **case 11** in the Reference book. This case yields the same end results, once we account for the actual dimensions and deflections, as follows. The length of the cantilever will be one half of the total length of the full beam, i.e. one half of the total height of the column, and also the deflection resulting from the relative push/pull between the two electronic components will be, for the cantilever beam, one half of that for the full column height.

I will use both methods here below to show what I am talking about.

Reference:

Machinery's Handbook, 16th Edition, 1962, The Industrial Press, New York. Library of Congress Card Number 14-2166.

Pages 387 through 395 give us the mathematical formulas for Stresses and Deflections in Beams.

Pages 358 through 361 give us the mathematical formulas for Moments of Inertia and Section Moduli of Sections.

Page 387: Definitions:

E = modulus of elasticity of the material;

I = moment of inertia of the cross-section of the beam;

Z = section modulus of the cross-section of the beam = I divided by the distance from neutral axis to extreme fiber;

W = load on beam;

s = stress in extreme fiber, or maximum stress in the cross-section considered, due to load W;

y = deflection measured from the position occupied if the load causing the deflection were removed.

L = Length of Beam

Note: All the above terms will be the same for both cases, i.e. the Washino and the Cherian cases, except for the stress "s", which we want to calculate and determine.

Pages 387 through 395: Stresses and Deflections in Beams.

Case 17. – (Beam) Fixed at One End, Free but Guided at the Other, with Load:

Stress at Support (pad) = $s = (W \times L) / (2 \times Z)$ (equation 2)

Maximum Deflection, at free end = $y = (W \times L^3) / (12 \times E \times I)$ (equation 3)

Case 11. – (Cantilever Beam) Fixed at One End, Load at Other:

Stress at Support (pad) = $s = (W \times L) / Z$ (equation 4)

Maximum Deflection, at end = $y = (W \times L^3) / (3 \times E \times I)$ (equation 5)

Page 358 through 361: Moments of Inertia and Section Moduli of Sections.

For a **Circular cross-section**, with a cross-sectional diameter "d":

Moment of Inertia = $I = (\pi \times d^4) / 64$ (equation 6)

Section Modulus = $Z = (\pi \times d^3) / 32$ (equation 7)

For a **Square cross-section**, with a side length "a":

Moment of Inertia = $I = (a^4) / 12$ (equation 8)

Section Modulus = $Z = (a^3) / 6$ (equation 9)

For a "**Rectangular**" cross-section, with a length/height of "d" and a width of "b", and **where "d" is in the direction of the bending, i.e. in the direction of the load W applied to the beam, i.e. the direction of the push and pull due to thermal deformation of the components, i.e. in the "radial" direction as defined above.**

Moment of Inertia = $I = (b \times d^3) / 12$ (equation 10)

Section Modulus = $Z = (b \times d^2) / 6$ (equation 11)

Let's first use **case 17**, which looks to be the more rigorous way. In our case, the "supports" of the beam are at the base of the joining means (solder joints or the like), which means at the attachment pads (pads). This is the spot where the stresses are the highest/worst.

We will calculate the highest stresses, which will occur at the pads, due to a certain deflection between the two pads, i.e. between the pads at each end of the joint (beam).

If the column/joint has a **circular** cross-section, then it would have no "orientation". The same shape of cross section will be under the same stress, regardless of how we rotate/orient the column. If the column/joint has a **square** cross section, then again, the orientation would not mean much, as long as the square has one side oriented radially. The important case to study and analyze is when the column/joint has a **rectangular** cross section. Then we would want to know if orienting the column/joint one way or another would make a difference in/on the stresses.

We will do so for two specific cases/orientations:

Orientation #1: Long Axis is Radial, which is Washino's radial orientation, and

Orientation #2: Short Axis is Radial, which is my (Cherian) transverse orientation.

But first let me do some work on the basic formulas themselves.

Let's start with Case 17.

The basic equations are:

Stress at Support (pad) = s = (W x L) / (2 x Z) (equation 2)

Maximum Deflection, at free end = y = (W x L^3) / (12 x E x I) (equation 3)

From equation 3, we can derive the following equation:

W = (y x 12 x E x I) / (L^3) (equation 12)

Substituting this value of W into equation 2, we get the following equation:

s = (W x L) / (2 x Z)

= { (W) x (L) } / (2 x Z)

= { (W) } x { (L) / (2 x Z) }

= { (y x 12 x E x I) / (L^3) } x { (L) / (2 x Z) } from equation 12

= { (y x 12 x E x I) x (L) } / { (L^3) x (2 x Z) }

= { (y x 12 x E x I) } / { (L^2) x (2 x Z) }

= { (y x 6 x E x I) } / { (L^2) x (Z) }

= { (6 x y x E x I) } / { (L^2) x (Z) } (equation 13)

Now, we will apply this equation 13 for the case of a beam with a rectangular cross section, and we will use the terms in equations 10 and 11 for the Moment of Inertia = I and the Section Modulus = Z, and substitute them into equation 13.

Moment of Inertia = I = (b x d^3) / 12 (equation 10)

Section Modulus = Z = (b x d^2) / 6 (equation 11)

s = { (6 x y x E x I) } / { (L^2) x (Z) } (equation 13)

$= \{ I / Z \} \times \{ (6 \times y \times E) \} / \{ (L^2) \}$

$= \{ [(b \times d^3) / 12] / Z \} \times \{ (6 \times y \times E) \} / \{ (L^2) \}$ from equation 10

$= \{ [(b \times d^3) / 12] / [(b \times d^2) / 6] \} \times \{ (6 \times y \times E) \} / \{ (L^2) \}$ from equation 11

$= \{ [(d) / 2] \} \times \{ (6 \times y \times E) \} / \{ (L^2) \}$

$= \{ [(d)] \} \times \{ (3 \times y \times E) \} / \{ (L^2) \}$

$= (d) \times \{ (3 \times y \times E) / (L^2) \}$ **(equation 14)**

We can simplify the looks of this equation, by combining the right hand side part of right side of the equation into one factor, by calling it K17, for Case 17, where

$K17 = (3 \times y \times E) / (L^2)$. (equation 15)

This is because **all the terms of this equation are identical in both cases,** i.e. the Washino's case and my Cherian case.

So, equation 14, becomes

$\boxed{s = (d) \times K17}$ **(equation 16)**

Now, let's work with **Case 11**, to see that we do get similar results:

Case 11. – (Cantilever Beam) Fixed at One End, Load at Other:

Stress at Support (pad) = $s = (W \times L) / Z$ (equation 4)

Maximum Deflection, at end = $y = (W \times L^3) / (3 \times E \times I)$ (equation 5)

Here, we will cut the column joint at its mid-height, and consider each half as a cantilever beam, fixed at its respective pad, and its free end being at the mid-height of the original column joint. This will also mean that the total push pull between the two components will be sensed at the mid-height will be only one half of the original amount.

To differentiate between the dimensions of the full height joint and the cantilever beam, we will use the following terminology:

L = Length of original joint

Lc = Length of Cantilever Beam = L/2

y = Deflection of the original joint

yc = Deflection of the Cantilever Beam = y/2

So, applying these terms into equations 4 and 5, we get:

Stress at Support (pad) = s = [W x (L/2)] / Z (equation 17)

Maximum Deflection, at end = y/2 = [W x $(L/2)^3$] / (3 x E x I) (equation 18)

Now, we repeat what we did earlier for Case 17;

From equation 20, we can derive the following equation:

W = { (y/2) x (3 x E x I) } / { $(L/2)^3$ }

= { (y/2) x (3 x E x I) } / { $(L)^3/(2)^3$ }

= { (y/2) x (3 x E x I) } / { $(L)^3/(8)$ }

= { (y) x (3 x E x I) } / { $(L)^3/(4)$ }

= { (12 x y x E x I) } / { $(L)^3$ } (equation 19)

Substituting with the value of W from equation 19, into equation 17, we get:

s = [W x (L/2)] / Z

= [W] x { (L/2) / Z }

= { (12 x y x E x I) } / { $(L)^3$ } x { (L/2) / Z }

= { (12 x y x E x I) x (L/2) } / { $(L)^3$ x Z }

= { (6 x y x E x I) x (L) } / { $(L)^3$ x Z }

= { (6 x y x E x I) } / { $(L)^2$ x Z }

= { [(6 x y x E) / $(L)^2$] } x { I / Z } (equation 20)

Substituting the terms for I & Z in equation 20, by the terms from equations 10 and 11, we get:

$\mathbf{s = \{ [(6 \times y \times E) / (L)^2] \} \times \{ I / Z \}}$ **(equation 20)**

$= \{ [(6 \times y \times E) / (L)^2] \} \times \{ [(b \times d^3) / 12] / Z \}$ (from equation 10)

$= \{ [(6 \times y \times E) / (L)^2] \} \times \{ [(b \times d^3) / 12] / [(b \times d^2) / 6] \}$ (from equation 11)

$= \{ [(6 \times y \times E) / (L)^2] \} \times \{ d / 2 \}$

$\mathbf{= \{ [(3 \times y \times E) / (L)^2] \} \times \{ d \}}$ **(equation 21)**

Again, we can simplify the looks of this equation, by combining the left hand side part of right side of the equation into one factor, by calling it K11, for Case 11, where

$\mathbf{K11 = (3 \times y \times E) / (L^2)}$**.** **(equation 22)**

This is because all the terms of this equation are identical in both cases, i.e. the Washino's case and my case.

So, equation 21, becomes

$\boxed{\mathbf{s = (d) \times K11}}$ **(equation 23)**

Equations 22 and 23 sure look identical to the above corresponding equations 15 and 16. Accordingly, the same analysis and conclusion, that will follow below, do apply to **both case 11** and **case 17 as well.**

ANALYSIS

Now, let us apply equation 16 or 23 to the two orientations of the elongated pads of Washino and mine.

In order to get a good feel for the difference between the stresses due to Washino and the stresses due to Cherian, let us give the pads some concrete dimensions. Let us say that the

pads are 0.010 inch wide and 0.050 inch long. Consequently, their long axis will be equal to 0.050 inch long, and their short axis will be equal to 0.010 inch long.

Let us also remember the following:

Each of **Washino's** elongated pads is oriented so that its "**long**" axis is "**radial**". This means that when we create a joint on top of such a pad and the cross section of the joint conforms substantially to the shape of the pad, the long axis of the joint's cross section, which is 0.050 inch long, will also be "radial". This means that the term "d" in equation 16 or 23 will be equal to 0.050 inch. Let's call this "d" value, the "d.Washino" or simply the "**dW**". So, dW = 0.050 inch.

In contrast, in my **Cherian** orientation, the "**short**" axis is "**radial**". This means that the term "d" in equation 16 or 23 will be equal to 0.010 inch. Let's call this "d" value, the "d.Cherian" or simply the "**dC**". So, dC = 0.010 inch.

If we now substitute the actual values of "d", i.e. dW and dC, in equation 16 or 23 for the two orientations, we will get the following stresses in the joints.

s (Washino) = sW = (dW) x K17

s (Washino) = sW = (0.050) x K17 (equation 24)

and

s (Cherian) = sC = (dC) x K17

s (Cherian) = sC = (0.010) x K17 (equation 25)

From equations 24 and 25, we can see that the stress with Washino's orientation **(equation 24)** is **five times higher** than with my (Cherian) orientation **(equation 25)**, i.e. **Washino's stresses are 500 percent higher than Cherian's stresses.**

Conversely, we can say that the stresses with my Cherian orientation are only 20% of the stresses with the Washino's orientation, or we can say that the **Cherian orientation has reduced the stresses by 80%.**

Of course, this depends on the dimensions we chose for the pads.

We can also describe the relation between Washino's and Cherian's stresses by the following "**ratio**" as follows:

sW / sC = { (dW) x K17} / { (dC) x K17}

= { (dW) } / { (dC) }

= (dW) / (dC)

= dW / dC

In other words,

sW / sC = { Length of Long Axis } / { Length of Short Axis } (equation 26)

So, regardless of the actual dimensions, we can see that **the ratio between Washino's stresses and Cherian's stresses is equal to the ratio between the length of the Long Axis divided by the length of the Short Axis of the elongated pads**. This is of course assuming that the pads in Washino and Cherian have the same dimensions and total area, and the only difference between them is their orientations.

So, if the Long Axis is 0.050 inch and the Short Axis is 0.010 inch, as above, then the ratio is **500 percent**. If we choose other dimensions, closer to each other, we would get a smaller increase in stress, but the ratio and the stresses will still be higher with Washino's orientation that with Cherian's orientation. **And the only reason is due to the orientation of the pads.**

CONCLUSION:

So, in summary and in conclusion, we can see that the **Cherian orientation** of the elongated pads has **dramatically reduced the stresses, simply by orienting the elongated pads**, such

that their "**Short** Axis" is "**radial**", **in contrast with Washino**'s orientation, which orients the "Long Axis" to be "radial".

In fact, the electronics industry will love my invention. The whole electronics industry has been struggling, for many years, with the problem that I am solving with my invention. The industry has been trying several methods, devices and approaches to solve the problem, but there is still a lot to be desired. My invention will help solve the problem.
And my solution is new and novel and nobody has thought about it until now, because it was not so obvious to anyone else before. And Washino definitely was not addressing this problem per se.

I am sure that Examiner would agree that this dramatic reduction in stresses in the joints, and especially at the pads, is a **technical and utility advantage**.

I also believe that this kind of technical and utility advantage was not known or envisioned by Washino and was **not obvious** to one having ordinary skill in the art at the time the invention was made, i.e. before my invention.

Accordingly, I would like to respectfully ask Examiner to remove his objection about the novelty and obviousness of my invention and to allow my claims.

Accordingly also, I believe that, based on my above explanations, my present newly amended claims should now be considered in condition for allowance and accordingly, I respectfully request the favor of allowing these claims, and of issuing a patent for them.

Thank you very much and best regards.

/Gabe Cherian/. Gabe Cherian